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Investigation of the Part-Load Performance of an Absorption Chiller

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ABSTRACT

An experimental investigation designed to determine the part-load performance of an ammonia-water absorption water chiller is described. The steady-state and cyclic performance of the chiller were measured under controlled conditions in an environmental chamber. Two valves were installed in the chiller to separate high- and low-pressure regions during off times, and insulation was applied to the chiller components. By these measures, losses due to cyclic operation were reduced by over 50%, resulting in a 6% to 7% increase in the calculated seasonal performance factor for typical northern and southern climates in the United States. The use of the valves eliminated the need of the "spindown" period, thereby reducing the consumption of parasitic electrical energy.

INTRODUCTION

An absorption water-chiller, like other building heating/cooling equipment, is generally selected to meet the design (i.e., maximum) load. Under design conditions, the water-chiller operates continuously at full capacity. During much of the cooling season, however, the load is less than the design value, and the water-chiller must be cycled on and off to provide the desired indoor conditions. The seasonal performance of the water-chiller is thus dependent on its performance during part-load operation.

Some information is already available in the literature on the performance of absorption water-chillers under part-load conditions.[1,6] The data available, however, demonstrate that the performance of an absorption air conditioner can be significantly degraded by cyclic operation. Reduced performance occurs for several reasons. When an absorption unit is operated, regions of high pressure (in the condenser and generator), and low pressure (in the absorber and evaporator) are established. When the unit is turned off, the pressure difference causes a migration of the working fluids from the high- to the low-pressure regions. This mass transfer is aided by temperature differences that exist between the components. In addition, radiative and convective heat transfer occurs between the individual components and between the components and the environment. When the unit is turned back on, energy must be expended to reestablish the high- and low-pressure regions, as well as to make up for the heat losses that occurred during the off period.

Manufacturers of absorption water-chillers recognize the problem with part-load operation and employ two methods to maintain acceptable performance during part-load conditions. The

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turbine meter to allow recording of the instantaneous and average flow during a test. The chilled water flow rate and inlet temperature were maintained constant at 0.45 l/s and 12.8°C (55°F) in all tests. Thermocouples were installed in the chilled water inlet and outlet streams close to the unit and were connected to the data acquisition system. In addition, two thermopiles were installed between the two streams and connected to a strip-chart recorder to provide a continuous record of the temperature difference. The estimated precision of the capacity measurement is within 1%.

The volumetric flow rate, temperature, and pressure of the natural gas were measured in each test. The higher heating value of the natural gas was determined in a calorimeter in operation at the National Bureau of Standards. The energy consumption of the combustor heating the generator was determined with an estimated precision of 2%.

Pressure gauges were mounted on the generator and absorber of the unit to measure the high- and low-side pressures. These pressures were recorded at the beginning and end of the burner on-time and spindown periods. In addition, for some cyclic tests, the pressures were recorded at short time intervals during the entire cycle. Both pressure gauges were calibrated on a deadweight gauge-tester prior to the tests.

Before most of the tests were conducted, two remote-activated ball valves were installed. Their positions within the unit are indicated in Fig. 1. In order to install the valves, it was necessary to remove ammonia and to replace it again after the work was completed. The ammonia charge was adjusted to maximize the steady-state capacity at the standard rating point (35°C [95°F] outdoor air temperature, 12.7°C [55°F] chilled water return temperature). This capacity was within 1% of that obtained before the valves were installed.

After testing the influence of the valves, the following parts of the absorption chiller were insulated with glass wool: the solution-cooled absorber, the receiver located before the solution pump, and the rectifier and that part of the generator that is not in direct contact with the burner. In addition, the panel between the components listed above and the condensing unit was insulated. To prevent further heat losses from the generator by convection, the exhaust for the flue gases was closed during off-times. The effect of the insulation on the steady-state performance was checked, and no change in the capacity was detectable.

TESTING PROCEDURE AND EVALUATION

All tests were conducted with a chilled water flow rate of 0.45 l/s (7.2 gpm) and a chilled water inlet temperature of 12.8°C (55°F). Prior to each steady-state test, steady conditions were first established in the environmental chamber and then data were taken and averaged over a 30-minute period. The steady-state capacity, Q_{ss}, was determined by the relationship

$$\dot{Q}_{ss} = \dot{m} C_p \Delta T_{ss} \tag{1}$$

where

$\dot{m}$ is the mass flow rate of chilled water, C_p is the specific heat of water, and ΔT_{ss} is the steady-state temperature difference between the inlet and outlet chilled water streams. Q_{ss} is the instantaneous capacity which is equal to the average capacity in steady-state operation

The steady-state coefficient of performance (COP_{ss}) is defined here as the ratio of the capacity to the sum of the rate of gas energy input to the combustors and the steady-state electrical power input to the blower and pumps, E_{ss}.

$$COP_{ss} = \frac{\dot{Q}_{ss}}{\dot{E}_{ss}} \tag{2}$$

This definition of COP weights the rate of gas energy input and power consumption equally. Since other equally appropriate definitions of COP exist, the experimental results in Tab. 1 contain the ratio of the electrical power to the rate of gas energy inputs.

strongly dependent on the air temperature, especially for temperatures higher than about 30°C (86°F). Increasing temperatures decreased the capacity significantly.

Part-Load Performance in the Original Operating Mode

Cycle rates were chosen for most of the tests according to the thermostat characteristics supplied by the manufacturer. (Thermostat cycle rates are not constant but rather a parabolic function of burner on-time.[3] The maximum recommended cycle rate for this absorption chiller is about 1.5 cycles per hour (CPH), which occurs at 50% on-time. At 20% and 80% on-time, the cycle rate with the recommended thermostat is 1.0 CPH.)

The tests listed in lines 5, 6, 7, 8, and 14 of Tab. 1 were conducted to measure the part-load performance of the absorption chiller in its original operating mode (i.e., without additional insulation, without the valves in operation, and with the spindown period enabled). The tests shown in lines 5 and 8 were conducted under the same conditions but at the beginning and in between the other tests, respectively, to check the reproducibility of the experimental data, which was found to be satisfactory. During a cooling season, an absorption chiller is operated at a variety of cycle rates and outdoor air temperatures. Tests were conducted to determine how changes in the outdoor temperature affect the part-load factor (PLF) at a given cycle rate. The results in lines 5, 6, and 7 of Tab. 1 show that both the part-load factor and the cooling load factor (CLF) decrease with increasing outdoor air temperature. The deviation in PLF between 21.6°C (71°F) and 35°C (95°F) is about 7%, while the change in CLF is about 9% In an installation in which the chiller is appropriately sized, part-load operation is more likely to occur at temperatures below the design condition. Therefore, all of the remaining cyclic tests were conducted at 26.6°C (80°F) as a representative condition.

The circles in Fig. 3 show the experimental values of PLF plotted versus CLF for the absorption chiller in its original operating mode over a range of part-load operating conditions. The size of the symbols in Fig. 3 is indicative of the uncertainty in the measured values. For comparison, the part-load performance of vapor compression systems (at a maximum cycle rate of 3 CPH), as assumed in Ref 4, is indicated by line a in Fig. 3. The performance of the absorption system is not considerably lower than that of vapor compression systems when it is operated at the recommended cycle rate, which is about one-half the cycle rate for vapor compression systems.

Part-Load Performance with the Values in Operation and without Insulation Applied

In this section, the part-load performance of the chiller in its original operating mode (lines 5, 6, 7, 8, and 14 of Tab. 1) is compared with its performance when the valves are closed during the burner off-time and open during the burner on-time (lines 10 and 16). The spindown period was disabled when the valves were operated. Tests were also conducted in which the valves were closed after the spindown period was completed. However, these tests resulted in slightly lower part-load factors than those with disabled spindown, and they required significantly more electrical energy. Apparently, the operation of the valves eliminates the need of the spindown period.

Comparing the temperature changes within the unit during the on-time, it was obvious that the average temperatures during the cycle are closer to their steady-state values when the valves are operated than when they are not. Further, when the valves were operated, there was no time delay for the temperature rise of the fluid leaving the solution-cooled absorber. This time-delay, which was 1.5 minutes in the original operating mode, indicates that the liquid absorbent solution traveled during the off-time from the generator to the solution-cooled absorber and needed to be pumped back to the generator. A similar time-delay was observed for the capacity in the original operating mode. Again, it took approximately 1.5 minutes after turning on the unit until it started to cool down the incoming water. This time-delay was not present when the valves were operated.

The valves also affect the behavior of the pressures as illustrated in Fig. 4. During the off-time, the high- and low-side pressures converge to the same pressure, and then they both drop with the same rate in the original operating mode. The low-side pressure remains unchanged during the off-time when the valves are operated, while the high-side pressure drops significantly in the beginning of the off-time but then stabilizes at a relatively high value. During the on-time, the high-side pressure achieves higher values (closer to the steady-state values) when the valves are operated, while the low-side pressure remains stable at its steady-state value. The peaks shown by the high- and low-side pressures after turning off the unit were observed in all cases in which spindown was disabled including those cases in

Seasonal Performance

The performance data obtained in this investigation were used to calculate the seasonal performance factor, SPF, of the absorption chiller. The seasonal performance factor is defined as the ratio of the total cooling load supplied to the total fuel and electrical energy consumed by the chiller during the cooling season. In the results given below, the fuel and electrical energy were equally weighted. To fairly compare absorption chillers with vapor compression machines, however, the electrical energy consumption should be divided by the efficiency of its generation for both systems.

The calculations were done for residential applications according to the modified bin-temperature method given in Ref 5. The modified bin-temperature calculation procedure uses the information in Fig. 3 to estimate the part-load performance of the chiller in each temperature bin and thereby provides an estimate of the seasonal performance including the effects of cyclic operation. The calculations were done for a generalized northern and southern climate;[5] the results are shown in Tab. 2.

Column 1 in Tab. 2 shows the values of SPF that would be achieved if the unit were to operate under all circumstances with the steady-state COP for any given air temperature. This is an upper limit for the SPF. Column 2 in Tab. 2 displays the SPF obtained by the unit in its original operating mode, while column 3 shows the SPF achieved by the unit with insulation in place, the valves in operation, and the spindown period disabled. Compared with the original operating mode, the insulation and valves increase the SPF by 7.0% in the southern climate and by 6.4% in the northern climate. Expressed in another way, these figures indicate that the losses due to part-load operation can be reduced by approximately 50% by the insulation and valves.

CONCLUSIONS

The results show that migration of the working fluids during the off-time is a major factor contributing to the degradation of the absorption chiller performance during cyclic operation. This migration can be reduced by the installation of automatic valves that separate the low- and high-pressure regions during the off-periods. The use of the valves eliminates the need for the spindown period and results in a significant reduction of electrical energy consumption during cyclic operation. Insulating the chiller components in this way also increased the part-load performance but not to the same extent as the valves. A 6% to 7% increase in the calculated seasonal performance factor results from the use of the valves and insulation.

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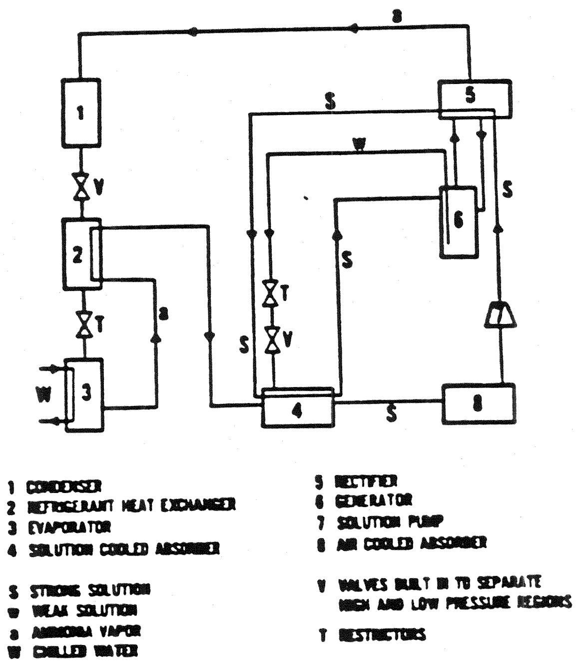

Figure 1. *Scheme of the absorption water chiller under investigation*

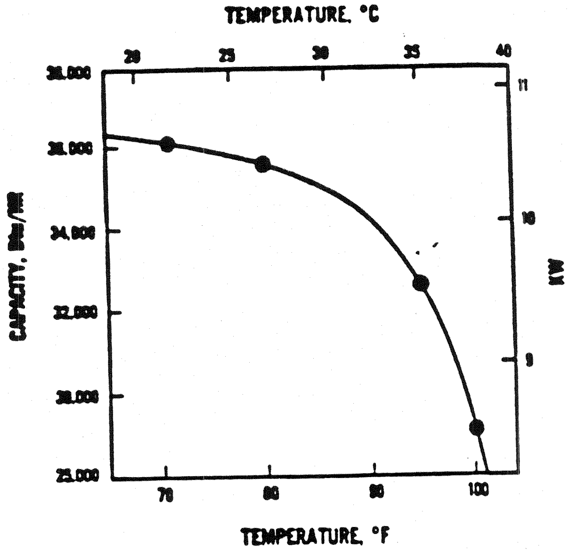

Figure 2. *Steady-state capacity of the absorption water chiller versus outdoor air temperature*

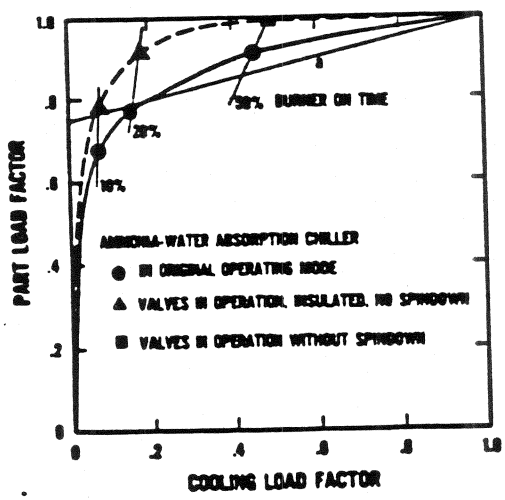

Figure 3. *Part load factor versus cooling-load factor for different cycle rates and operating modes of the absorption chiller*

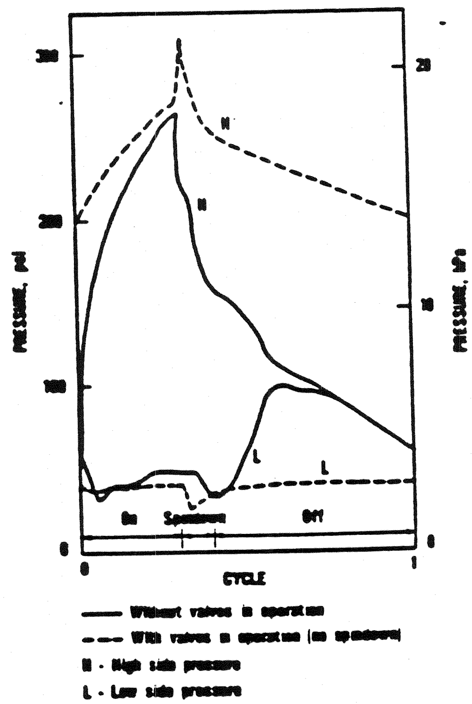

Figure 4. *Comparison of typical behavior of high- and low-side pressures during a cycle with and without valves in operation; spindown was enabled in the latter case*